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TECHNICAL TRANSLATION

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A VIBRATION ABSORBER FOR TWO-BLADED HELICOPTERS

By Th. Laufer

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A VIBRATION ABSORBER FOR TWO-BLADED HELICOPTERS[*]

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I. INTRODUCTION

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We recognize the advantages of a two-bladed rotary wing configuration, such as: hub simplicity, absence of drag hinge dampers, ruggedness, et cetera. For the jet helicopter which utilizes the blades as air ducts, there is the added advantage for a given section of passage, a smaller plenum and, consequently, a higher ratio of take-off weight to necessary ejector thrust. These advantages are opposed by the less favorable vibrating behavior, because the 2nd harmonic of the blade motion is not in equilibrium. One is, therefore, led to anticipate an isolation of the rotor by elastic suspension.

The object of this article is the study of the possibility of using a dynamic absorber. The principle is known. To balance the system, one adds another system which has the same natural frequency as that of the exciting force. We could then conceive of a FRAHM type antivibrator mounted on the pylon of the helicopter and given a frequency 2ω = const. (ω = frequency of rotor).

One such absorber would have, in addition to the necessarily large mass, the defect of only operating correctly at a well-defined frequency. On the other hand, at some frequencies slightly below or above, the amplitudes would be cumulative. Now, even for an engine-driven helicopter the frequency of the rotor is not absolutely constant, which is all the more reason for the jet helicopter where the possibilities of variation of the rpm present an essential attraction to the system.

It is necessary, consequently, to choose an absorber based on the utilization of the centrifugal force as in the case of the Taylor pendulum. But whereas the latter absorbs the torsional vibrations of a crankshaft, the present antivibrator must cancel the rotating forces at 2ω in the horizontal plane and the oscillating forces in the vertical plane at 2ω.

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[*]Translation of "Un Étouffeur de vibrations pour hélicoptère bipale." Technique et Science Aéronautiques, No. 4, Aug. 1959, pp. 231-235.

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2. VERTICAL OSCILLATIONS

Recall first the principle of an absorber using as the elastic force, the centrifugal force. Figure 1 shows the schematic of one such absorber of vertical vibrations. The blades are in a normal position to the design plane and are attached to the hub by flapping hinges. The motion of the hub is not directly related to the cyclic pitch changes of the blades; thus, these changes and the general pitch of the blade are effected in the classic manner by a spider control which is not illustrated in figure 1. The hub has two arms which form a rigid assembly. Each of the arms supports a hinged mass at its extremity.

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The arms form two flapping pendulums in the field of the centrifugal force. The higher the rpm, the greater the force which restores the pendulum to its equilibrium position; the more the pendulum stiffens, the more the natural frequency increases. One sees immediately (eqs. of fig. 1) that the length of the pendulums must be one-fourth the distance of the masses to the center of the rotor in order that its own natural frequency will always be equal to twice that of the rotor rpm. Let us note, however, that some measures of acceleration in flight have shown that the vertical component of vibration is generally low, so that some machines, such as the Djinn, may not even include an isolator.

The application of this apparatus does not seem to pose a problem, neither in descending vertical flight nor in translational flight where the disturbing forces seem to be weak.

On the other hand, in the case of the free hub, its stabilization necessitates, in any case, an increase of inertia in the indicated direction, so that the realization of such an absorber - for the case where it proves itself applicable - would consist of a simple addition of two hinges on the horizontal axis.

3. HORIZONTAL OSCILLATIONS

Since the aerodynamic conditions for the advancing blade are different than for the retreating blade, there is introduced in the elastic system, formed by the blades and the hub with its suspension to the fuselage, an exciting force of which the horizontal component is most troublesome at a frequency of 2ω.

As we propose to balance - by an absorber mounted on the rotating system - the rotor, where the reaction is in the horizontal plane, the natural frequency must necessarily be ω. The phenomena are complicated by the elastic deformation of the blade and elastic suspension of the hub. Furthermore, take-off poses the problem of stability known as ground resonance.

We will first roughly describe the function of such an absorber; then we will summarize in a few tables the complete calculation.

Figure 2 shows the schematic of such an absorber applied to a Djinn-type rotor. The two blades are attached together by packs of leaf springs, and as in the preceding explanation, receive their cyclic change independently of the hub. Two arms are rigidly attached to the hub and support two pendulums that can flap in the plane of the hub.

In order to explain schematically the operation of the blades, let us recall first that owing to their elasticity, they are allowed to bend in the horizontal plane, even if unhinged. In the present example, the bending takes place due to their elasticity and that of the packs of leaf springs. Let us look at the blade motion in two characteristic positions, as shown in figure 3. In figure 3(a), the blades are at their maximum amplitude a little after the maximum excitation by the aerodynamic forces. The centrifugal forces of the blades then give a resultant in the plane of symmetry.

Figure 3(b) shows the blades 90° later. In order to come from the advancing position to the retreating position, the blades must describe an angle greater than 180° and will have, consequently, a speed greater than that of the average rotor speed.

It follows that the centrifugal forces will be greater than for that of the other blade, which will have a speed lower than the mean rpm. The resultant has then changed sign in comparison to that found for the position which preceded it by 90°. Thus we have a motion having a frequency twice that of the rotor rpm.

The figures 3(a) and 3(b) show the starting positions of the two pendulums of the dynamic absorber. Like the blades, the advancing pendulum in forward flight must traverse a greater angle than the retreating pendulum and have, consequently, a greater angular velocity and centrifugal force. This difference of centrifugal forces serves to balance the resultant of the centrifugal forces of the two blades; in an analogous manner, when the blades are alined but have a different speed (fig. 3(b)), it is the resultant of the two centrifugal forces of the absorber inclined in relation to each other which will establish equilibrium.

4. CALCULATION OF THE HORIZONTAL ABSORBER

After this simplified account of the operation, we give a resumé of the calculation.

4.1. Method and Hypotheses

The hypotheses are those which have been made by Feingold and Coleman in their study "Theory of Mechanical Oscillations of Rotors With Two Hinged Blades" (ref. 1).

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The amplitudes are assumed small. The elastic deformation of the blades is replaced by the motion of a rigid articulated blade provided with a restoring spring. The aerodynamic damping is neglected. One takes into account the effect of the mass of the fuselage, of the elasticity and damping of the elastic suspension of the hub or of the landing gear during descending vertical flight, in giving to the hub its corresponding characteristics.

The mass of the stabilizer is assumed to be concentrated at the center of gravity of the small masses of the pendulums. We will utilize the Lagrange equations.

Figure 4 gives the geometrical symbols which have been used.

The following section 4.2. defines the symbols which have been used.

4.2. Symbols

x_1, y_1, x_2, y_2 coordinates of C. G. of the blades in the fixed system

$\bar{x}_1, \bar{y}_1, \bar{x}_2, \bar{y}_2$ coordinates of the small masses of the stabilizer in the fixed system

β_1, β_2 horizontal flapping of blades

$\bar{\beta}_1, \bar{\beta}_2$ horizontal flapping of the stabilizer masses

x, y hub coordinates in the rotating system

ω angular velocity of rotor

t time

a	distance of hinge to center of rotation
$\bar{a}$	distance of stabilizer arm hinges to the center of rotation
b	distance of blade C. G. to hinge
$\bar{b}$	distance of stabilizer mass C. G. to hinge
m_b	mass of blade
m_e	mass of stabilizer tip masses
m	effective mass of hub
K_β	coefficient of elasticity of blade about hinge
K	coefficient of elasticity of the suspension and of the landing gear concentrated in the hub
B	coefficient of damping of the suspension and of the landing gear concentrated in the hub

$$M = 2m_b + 2m_e + m$$

$$\theta_0 = \frac{b}{2}(\beta_1 + \beta_2), \quad \frac{a}{b} = \alpha, \quad \mu_e = \frac{2m_e}{M}$$

$$\theta_1 = \frac{b}{2}(\beta_1 - \beta_2), \quad \frac{\bar{a}}{b} = \bar{\alpha}, \quad \mu_b = \frac{2m_b}{M}$$

$$\bar{\theta}_0 = \frac{\bar{b}}{2}(\bar{\beta}_1 + \bar{\beta}_2), \quad 1 + \frac{r^2}{b^2} = \rho$$

$$\bar{\theta}_1 = \frac{\bar{b}}{2}(\bar{\beta}_1 - \bar{\beta}_2)$$

$\nu_f = \sqrt{\frac{K}{M}}$ natural frequency of the effective hub

ν_b natural frequency of the hub

$$\lambda = \frac{B}{M}$$

r	radius of gyration of the blade about its C. G.
S_a	distance from the center of the blade to the hinge

$\bar{S}_a$	distance from the center of the stabilizer to its hinge
A_1	aircraft weight
T	kinetic energy
V	potential energy
F	dissipated energy
D_{rot}	aerodynamic drag of rotor
D_{max}	aerodynamic drag of hub
D_{eq}	aerodynamic drag of stabilizer

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4.3. Energies of the System

The energies are first expressed in the fixed system x_f, y_f. Then we transform these expressions into a rotating system at a speed ω with the rotor (compare to the Feingold report (ref. 1)).

<u>4.3.1. Kinetic energy</u>.- To the energy terms, in the manner Feingold has established, then, are added the terms of the kinetic energy of the masses of the stabilizer expressed in the fixed system.

If we do not drop the terms which will occur in the Lagrange equations, the expression of kinetic energy of the system, expressed in the variables of the rotating system, is given by the following expressions:

$$T = \frac{M}{2}\left[\dot{x}^2 + \dot{y}^2 + \omega^2\left(x^2 + y^2\right) - 2\omega(\dot{x}y - \dot{y}x)\right] + m_b\left[\left(\dot{\theta}_0{}^2 + \dot{\theta}_1{}^2\right)\left(1 + \frac{r^2}{b^2}\right)\right.$$

$$\left. + 2\dot{y}\dot{\theta}_1 + 2x\omega\dot{\theta}_1 - 2\dot{x}\omega\theta_1 + 2y\omega^2\theta_1 - \frac{a}{b}\,\omega^2\left(\theta_0{}^2 + \theta_1{}^2\right)\right]$$

$$+ m_e\left[\dot{\bar{\theta}}_0{}^2 + \dot{\bar{\theta}}_1{}^2 - 2\bar{x}\dot{\bar{\theta}}_1 + 2\bar{y}\omega\dot{\bar{\theta}}_1 - 2\omega\bar{\theta}_1 - 2\bar{x}\omega^2\bar{\theta}_1 - \frac{\bar{a}}{\bar{b}}\,\omega^2\left(\bar{\theta}_1{}^2 + \bar{\theta}_1{}^2\right)\right]$$

or

$$\theta_0 = \frac{b}{2}\left(\beta_1 + \beta_2\right),\ \bar{\theta}_0 = \frac{\bar{b}}{2}\left(\bar{\beta}_1 + \bar{\beta}_2\right)$$

$$\theta_1 = \frac{b}{2}\left(\beta_1 - \beta_2\right),\ \bar{\theta}_1 = \frac{\bar{b}}{2}\left(\bar{\beta}_1 - \bar{\beta}_2\right)$$

and

$$M = 2m_b + 2m_e + m$$

4.3.2. Potential energy.- The potential energy due to the elastic deformations of the blades and of the undercarriage is:

$$V = \frac{1}{2} K\beta\left(\beta_1{}^2 + \beta_2{}^2\right) + \frac{1}{2} K\left(x^2 + y^2\right)$$

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4.3.3. Dissipation of energy.- The dissipation of energy due to the whole undercarriage suspension is:

$$F = \frac{1}{2} \beta\left[\dot{x}^2 + \dot{y}^2 + \omega^2\left(x^2 + y^2\right) - 2\omega(\dot{x}y - x\dot{y})\right]$$

4.4. Exciting Forces

In forward flight, the advancing blade encounters aerodynamic conditions which differ from those on the retreating blade. An excitation of 2ω is introduced in a fixed system and ω for a rotating system. If we neglect the harmonics and choose the axis $f\rho$ of the fixed reference in the position of the component of the resultant average aerodynamic rotating drag D_{rot} on the rotor plane, we obtain the forces given by the following expressions:

Blades:

$$-\frac{S_a}{b} D_{rot} \sin \omega t \quad \text{for} \quad \theta_1 = \frac{b}{2}\left(\beta_1 - \beta_2\right)$$

Absorber:

$$-D_{eq} \frac{\bar{S}_a}{\bar{b}} \cos \omega t$$

Hub:

For x: $D_{max} \cos \omega t$

For y: $-D_{max} \sin \omega t$

$$\theta_1 = \frac{b}{2}\left(\beta_1 - \beta_2\right)$$

Remembering that for $\theta_0 = \frac{b}{2}(\beta_1 + \beta_2)$, which represents the motion in phase of the two blades, the motion is not susceptible to coupling with the pylon and thus the corresponding exciting forces do not enter into the problem.

4.5. Lagrange Equations

Writing the Lagrange equations

$$\frac{d}{dT}\left(\frac{\partial T}{\partial q_r}\right) - \frac{\partial q_r}{\partial T} + \frac{\partial V}{\partial q_r} + \frac{\partial F}{\partial q_r} = D$$

with the energies T, V, F, and the exciting forces D of the preceding paragraphs, we arrive at a system of equations of motion:

$$D_{max} \cos \omega t = M(\ddot{x} - \omega\dot{y}) - 2m_b\omega\dot{\theta}_1 - 2m_e\ddot{\bar{\theta}}_1 - M(\omega^2 x + \omega\dot{y}) - 2m_b\omega\dot{\theta}_1 + 2m_e\omega^2\bar{\theta}_1$$

$$+ Kx + \beta\dot{x} - \beta\omega y - D_{max} \sin \omega t$$

$$= M(\ddot{y} + \omega\dot{x}) + 2m_b\ddot{\theta}_1 - 2m_e\omega\ddot{\theta}_1 - M(\omega^2 y - \omega\dot{x}) - 2m_b\omega^2\theta_1$$

$$- 2\omega m_e\dot{\bar{\theta}}_1 + Ky + B\dot{y} + B\omega x$$

$$\frac{S_a}{b} A_0 = 2m_b\left(1 + \frac{r^2}{b^2}\right)\ddot{\theta}_0 - 2m_b \frac{a}{b} \omega^2\theta_0 + \frac{2K\beta}{b^2} \theta_0$$

$$0 = 2m_e \frac{a}{b} \omega^2\bar{\theta}_0 + 2m_e\ddot{\bar{\theta}}_0$$

$$\frac{S_a}{b} A_1 \sin \omega t = 2m_b\left[\left(1 + \frac{r^2}{b^2}\right)\ddot{\theta}_1 + \ddot{y} + \dot{x}\omega\right]$$

$$- m_b\left[-2x\omega + 2y\omega^2 + 2 \frac{a}{b} \omega^2\theta_1\right] + 2 \frac{K\beta}{b^2} \theta_1$$

$$-D_{eq}\,\frac{\bar{S}_a}{\bar{b}}\cos\omega t = 2m_e\left(-\ddot{\bar{x}} + \dot{\bar{y}}\omega + \bar{\theta}_1\right) + 2m_e\left(\dot{\bar{y}}\omega + \bar{x}\omega^2 + \frac{\bar{a}}{\bar{b}}\,\omega^2\bar{\theta}_1\right)$$

The third and fourth of these equations contain only one variable each; that is to say, θ_0 and $\bar{\theta}_0$, and so may be studied outside of the system. The third equation corresponds to a damped motion of the blades. The fourth equation corresponds to a theoretical undamped motion of the stabilizer. But, as there is always some friction, for practical purposes it is a matter of a damped motion. The essential thing is that the two motions are not coupled with that of the hub and do not represent a danger of instability. It is where the motion of the two opposite arms are in phase that one encounters the greatest danger of instability.

4.6. Solution of the System of Equations

Seeing the complexity of the system of equations, the algebraic solution is not of interest but it is desirable to seek the numerical solution directly by finding:

(1) The proper values for the whole range of rpm (T. O. included)

(2) The response in the permanent operating regime

These calculations, which amount to the inversion of a matrix, will be preferably executed by electronic calculating machines.

5. RESULTS

Let us say at once that the results presented are purely theoretical, as no practical application of this absorber has been effective to date.

The equations were applied to a single machine, namely, the Djinn.

5.1. Proper Values

As was to be expected, the introduction of a system in which the natural frequency is equal to the operating rpm, introduces an instability as in the case when the natural frequency of the blade in the horizontal plane happens to be in this zone. Let us note that, in the case of a blade suspended by packs of leaf springs (ref. 2), this has much less importance for a new construction. In this case, one adapts the distance between the axis of the two packs in such a way as to

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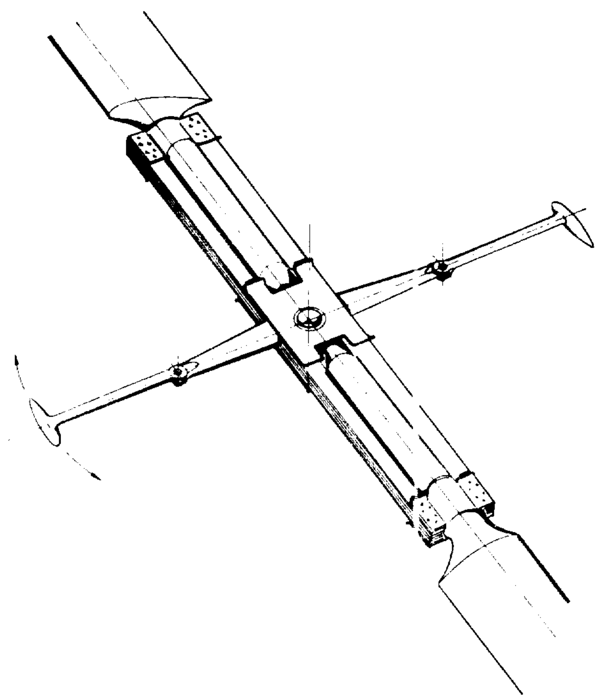

Figure 2.- Horizontal absorber.

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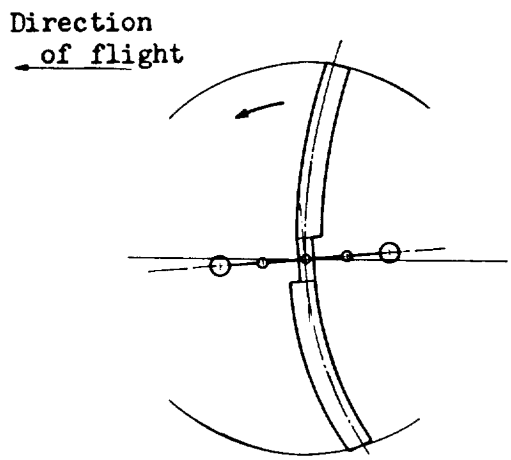
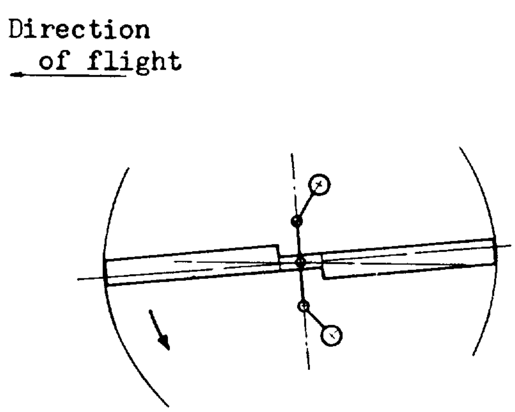


(a)



(b)

Figures 3(a) and 3(b).- Working schematic of a horizontal absorber.

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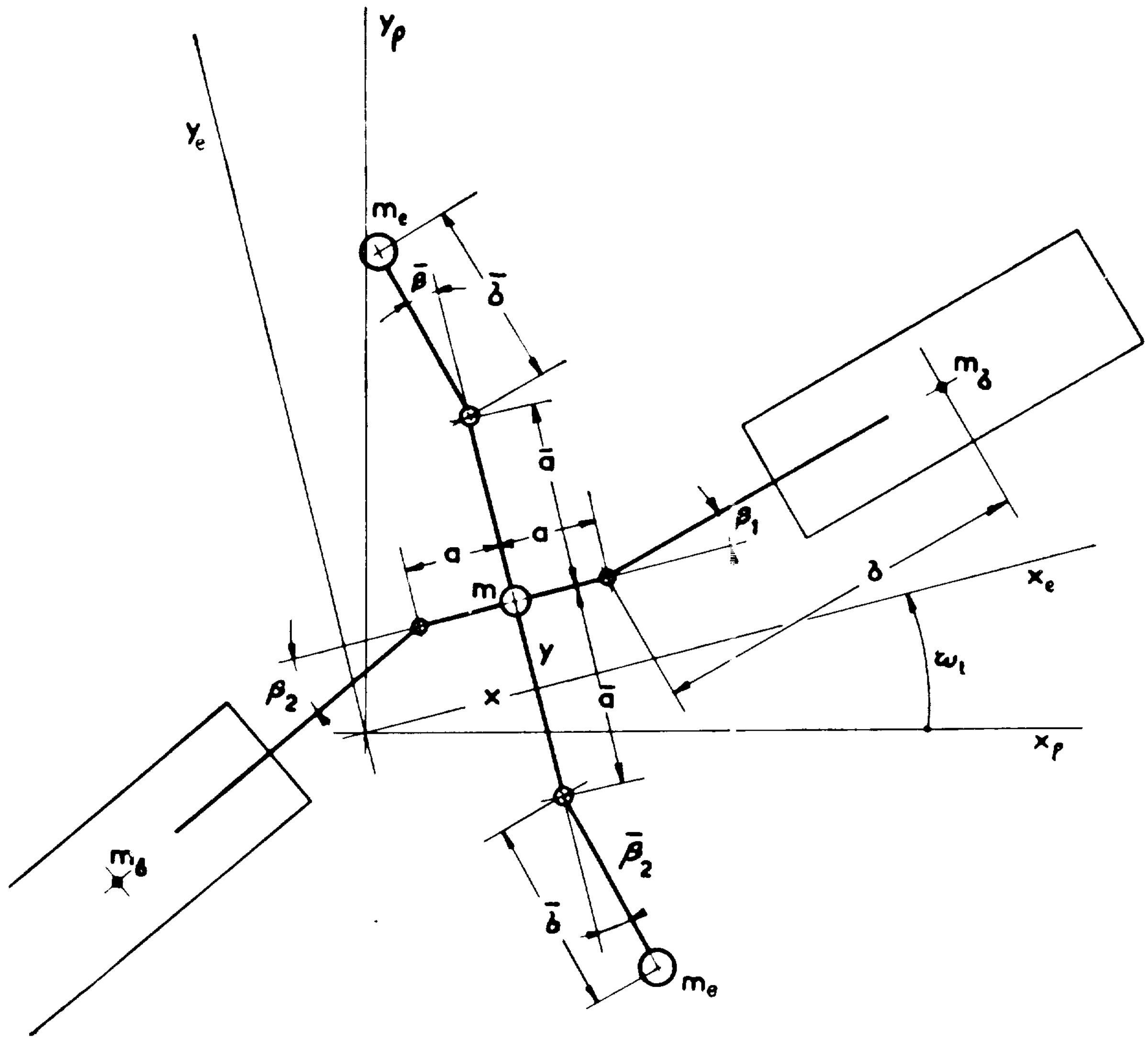

Figure 4.- Geometric symbols.